Heat exchanger and plate used in a heat exchanger

The invention relates to a heat exchanger, in particular an oil cooler for a vehicle, equipped with several tray-shaped plates which are placed on top of one another, are sealed together on their peripheral edge and are provided with passages, where passages lying essentially above one another form a continuous flow channel that traverses the plates. The invention further relates to a particularly suitable plate for a heat exchanger.

A heat exchanger of this kind, also called a stacked-plate heat exchanger, is known for example from DE 100 49 890 A1. In the stacked construction, metal plates of a trough-shaped design are soldered directly together at their peripheral edges. The plates have the same or identical shape, such that the number of necessary components is kept low. The heat transfer surface is determined by the number of plates and, as a result, by the length of the flow channel and by the dimensions of the flow channel itself. The greater the number of plates and the sizes of the flow channel, the greater therefore is the heat transfer surface, with at the same time a decreasing Reynolds' number. Effective heat exchange is thus limited because, with a maximum number of plates, an increase in heat exchange, afforded by the advantage of a greater heat transfer surface, can no longer be achieved because of the disadvantage of a smaller heat exchange on account of the lower Reynolds' number. In addition, the production costs are the higher the more plates are used.

The object of the invention is therefore to make available a heat exchanger which permits an increase in the heat exchange while having essentially the same or similar external dimensions of the heat exchanger and good utilization of the heat transfer surface.

According to the invention, this object is achieved by a heat exchanger of the type mentioned in the introduction and having the features of claim 1.

The invention is based on the concept that a more intensive heat exchange should be permitted while retaining as far as possible the structural design, i.e. the dimensions, in particular the external dimensions, of the heat exchanger. An objective is to ensure that a structural adaptation of the heat exchanger cancels the contradictory criteria - increase in heat transfer surface with decreasing Reynolds' number - such that the reynolds' number as far as possible does not decrease. For this purpose, a heat exchanger with several tray-shaped plates provided with passages is geometrically simplified in that a flow channel which is formed by passages lying essentially above one another and which traverses the plates, has an elongate cross section. By means of such a simple geometric change to the heat exchanger, it is possible, while retaining the same structural volume of the heat exchanger, to ensure a more intensive cooling by greater heat transfer, without the Reynolds' number decreasing.

In a preferred embodiment, the respective flow channel has an oval or rectangular cross section. This affords an advantageous utilization of space.

Different flow channels, in particular flow channels lying adjacent to one another, can expediently have different cross-sectional shapes. For example, a flow channel designed as an admission line can have an oval cross section and a flow channel designed as a discharge line can have a rectangular cross section. Similarly, a flow channel for a first medium can have a more elongate cross section than a flow channel for a second medium. Depending on the nature and design of

the heat exchanger, the flow channels can traverse the heat exchanger in different directions rectilinearly and/or in loops with and/or without reversal of their direction.

The cross section of a flow channel preferably has a length to width ratio L/B of between 1.5 and 12, preferably between 1.5 and 6, where L is a length and B is a width of the flow channel cross section. Especially in the case of small heat exchangers, for example motor vehicle oil coolers with 15 mm <= L <= 25 mm, the length to width ratio L/B is particularly preferably between 1.5 and 3 or, especially in larger heat exchangers such as industrial coolers with 50 mm <= L <= 80 mm, between 4 and 6.

The heat exchanger is particularly suitable for use as a stacked-plate cooler, in particular a stacked-plate oil cooler for a vehicle. The respective plates for a heat exchanger of this kind are of essentially identical design and in their simplest form have passages which are arranged next to one another and which have a substantially elongate cross section, for example a rectangular or oval cross section or a dome-shaped cross section.

Illustrative embodiments of the invention are explained in more detail below with reference to a drawing, in which:

Figure 1 shows a schematic representation of a heat exchanger, in particular a stacked-plate heat exchanger with flow channels,

Figure 2 shows a schematic representation of an embodiment for a plate of a heat exchanger a) according to the prior art and b) according to the present invention,

Figure 3 shows a diagram depicting the profile of the specific heat output Q/dTe as a function of the flow volume over time V/t of the media flowing through the heat exchanger,

Figure 4 shows a schematic representation of a connector element for a heat exchanger according to Figure 1.

Parts corresponding to one another are provided with the same reference labels in all of the figures.

Figure 1 shows a heat exchanger 1 which is used, for example, as oil cooler in a vehicle for a combustion engine. The heat exchanger 1 is designed as a stacked-plate heat exchanger. For this purpose, the heat exchanger 1 comprises several and in particular tray-shaped plates 2a to 2z (hereinafter referred to simply as plates 2). The plates 2 are stacked or placed on top of one another and sealed together at their peripheral edges, e.g. soldered. The plates 2 are provided with passages 4. The plates 2 are of essentially identical design. The passages 4 are provided as far as possible at the same positions above one another, such that, when the plates 2 are stacked on top of one another, a flow channel 6 is formed through the passages 4 lying above one another. The passages 4 lying above one another in the plates 2 thus have substantially identical dimensions and cross-sectional shapes. Passages 4 arranged adjacent to one another and forming several separate flow channels 6 can have other dimensions and other cross-sectional shapes. The respective shape and length of the flow channel 6 is determined in particular by a medium M flowing through the flow channel 6.

As is shown in Figure 1 in a possible embodiment for a heat exchanger 1, a first flow channel 6a is traversed by a first medium M1 in flow direction R1. The first

flow channel 6a serves here as an admission channel or admission line from which, along the respective plate 2, the first medium M1 flows to an opposite second flow channel 6b designed as a collecting channel and is removed again there in reverse flow direction R2 from the heat exchanger 1.

The first medium M1 is, for example, an engine oil that is to be cooled. The first medium M1 is admitted and removed via an admission pipe 8a and discharge pipe 10a, respectively, which in the illustrative embodiment are arranged on the top face of the heat exchanger 1. Depending on the nature and design of the heat exchanger 1, the admission and discharge can also take place on the underside of the heat exchanger 1 or on another side or on separate sides.

The second medium M2 is a coolant which is fed to and removed from the heat exchanger 1 via associated admission pipes 8b and discharge pipes 10b, respectively, for the purpose of cooling the oil. To allow the second medium M2 to flow through the heat exchanger 1 in flow direction R3, the respective plates 2 have further passages 4 which form further flow channels 6c and 6d. The coolant flows, in an analogous manner to the oil, through the associated flow channel 6c with reversal of the flow direction R3 into a flow direction R4 and/or without reversal (not shown).

For best possible heat transfer, the respective flow channels 6a, 6b, 6c, 6d have an elongate cross section QS. The cross section QS is preferably rectangular or oval. Flow channels 6a, 6b, 6c and/or 6d lying adjacent to one another, and thus the associated passages 4, can have different cross sections. The respective flow channel 6a, 6b, 6c and/or 6d preferably has in cross section a length l of 10 mm to 20 mm and a width b of 5 mm to 10 mm.

One of the plates 2 is shown in detail in Figure 2b. The plate 2 has four passages 4 which, by stacking of several plates 2 above one another, form one of the flow channels 6a to 6d. As a result of the elongate cross section QS - rectangular or oval - of the passages 4, it is possible, while retaining the external dimensions of the heat exchanger 1 in relation to a conventional heat exchanger with round passages as shown in Figure 2a, to increase the heat transfer surface A that extends between the passages 4. The areas between the passages 4 and the edge of the plate 2 contribute only to a small extent to a heat transfer and are therefore not included here in the heat transfer surface A.

The change in size of the cooling surface of the heat exchanger 1 according to the invention compared to a conventional heat exchanger is presented below on the basis of a given example, with identical dimensions of the two heat exchangers:

	surface
Heat exchanger according to the prior art	6384 mm^2
Heat exchanger according to the invention	7600 mm^2

By increasing the cooling surface with an elongate cross section QS for the passages 4 of the flow channels 6a to 6d, an increase in the specific heat output Q/dTe as a function of volume throughput Qv is achieved. A comparison of the change in the specific heat output Q/dTe of a conventional heat exchanger and the heat exchanger 1 according to the invention is set out below. Here, the specific heat output Q/dTe is the heat output normalized to a temperature difference Te at the cooler inlet. Moreover, the volume throughput Qv is defined as the flow volume V of the medium M1 or M2 flowing through the respective flow channel 6a to 6d in the time t.

Figure 3 shows a diagram depicting the profile of the specific heat output Q/dTe as a function of the flow volume over time V1/t of the medium M1 flowing through the heat exchanger 1, in the heat exchanger 1 according to the invention (measurement points with solid connecting lines) and according to the prior art (measurement points with interrupted connecting lines), in each case for different fixed flow volumes over time V2/t of the respective other medium M2 flowing through the heat exchanger. It will be seen from Fig. 3 that, by increasing the heat transfer surface A according to the present invention, it is possible, in a heat exchanger type selected by way of example, to achieve an increase in the specific heat output of up to approximately 20%.

Figure 4 shows an example of a possible embodiment for a connector element 12 which is adapted to the changed cross section QS of the respective flow channel 6a to 6d of the heat exchanger 1. Here, the connector element 12, on the side directed toward the heat exchanger 1, also has an elongate cross-sectional shape and, on the opposite side, the connector element 12 has for example a round cross-sectional shape for attachment of lines or tubes for the admission and/or discharge of the first medium M1 and/or of the second medium M2.

List of reference labels

1	heat exchanger
2	plates
4	passage
6a to 6d	flow channel
8a, 8b	admission pipe
10a, 10b	discharge pipe
12	connector element
b	width of a passage
dP1	pressure loss for medium M1
dP2	pressure loss for medium M2
dTe	temperature difference
l	length of a passage
M1	first medium
M2	second medium
Q	heat transfer quantity
QS	cross section
Qv	flow volume
R1 to R4	flow direction